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(54) Planetary range gear

(57) A speed range transmission is driven from the output shaft 56 of a multiple speed ratio, manual, synchronized power transmission and produces four-speed ranges including reverse for each speed ratio of the synchronized transmission. The range transmission includes a compound planetary gear set having forward planetary pinions 104 continuously engaged with a sun gear 100 formed on the output shaft 56 of the synchronized transmission. A rearward planetary pinion 106 is formed as a unit with each forward planet pinion. A first shift fork 130 is selectively engageable with splines 116 or 131 to lock against rotation either the planetary carrier 114 or a forward ring gear 128 which is continuously engaged with the first planetary pinion. The planetary carrier 114 and a rearward ring gear 140 engaged with the second planetary pinion are selectively engageable with a clutch sleeve 146 movable axially by a second shift fork 164. As the shift forks 130, 164 are moved axially, the clutch sleeve is selectively connected driveably to the rear ring gear or to the planetary carrier for forward range drive or to both of these in order to lock up the planetary gear set to the output shaft 118 for a high speed range drive condition.

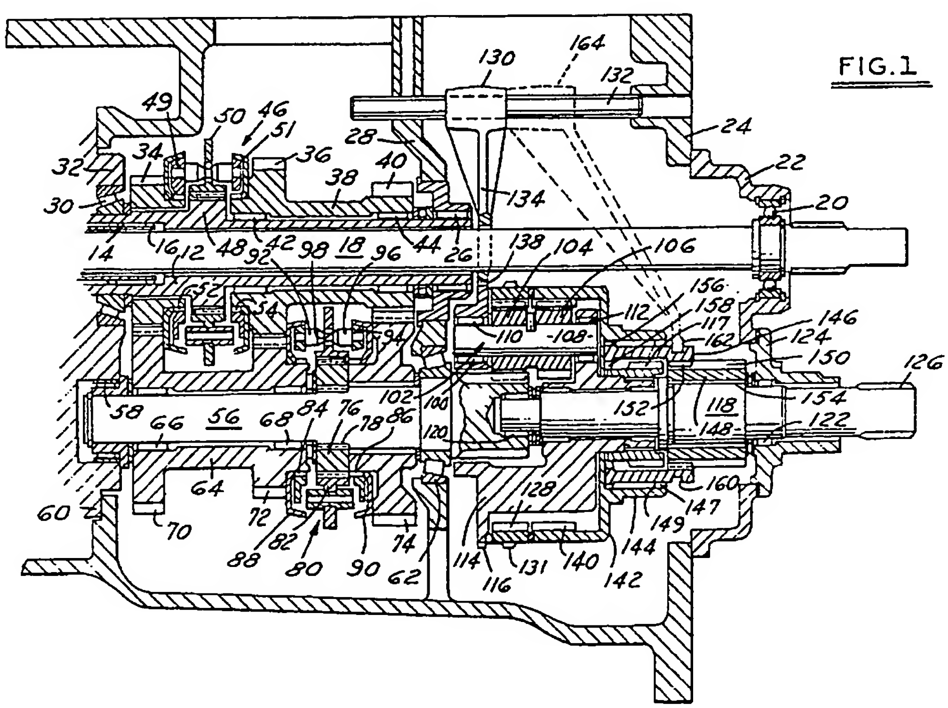

FIG.1

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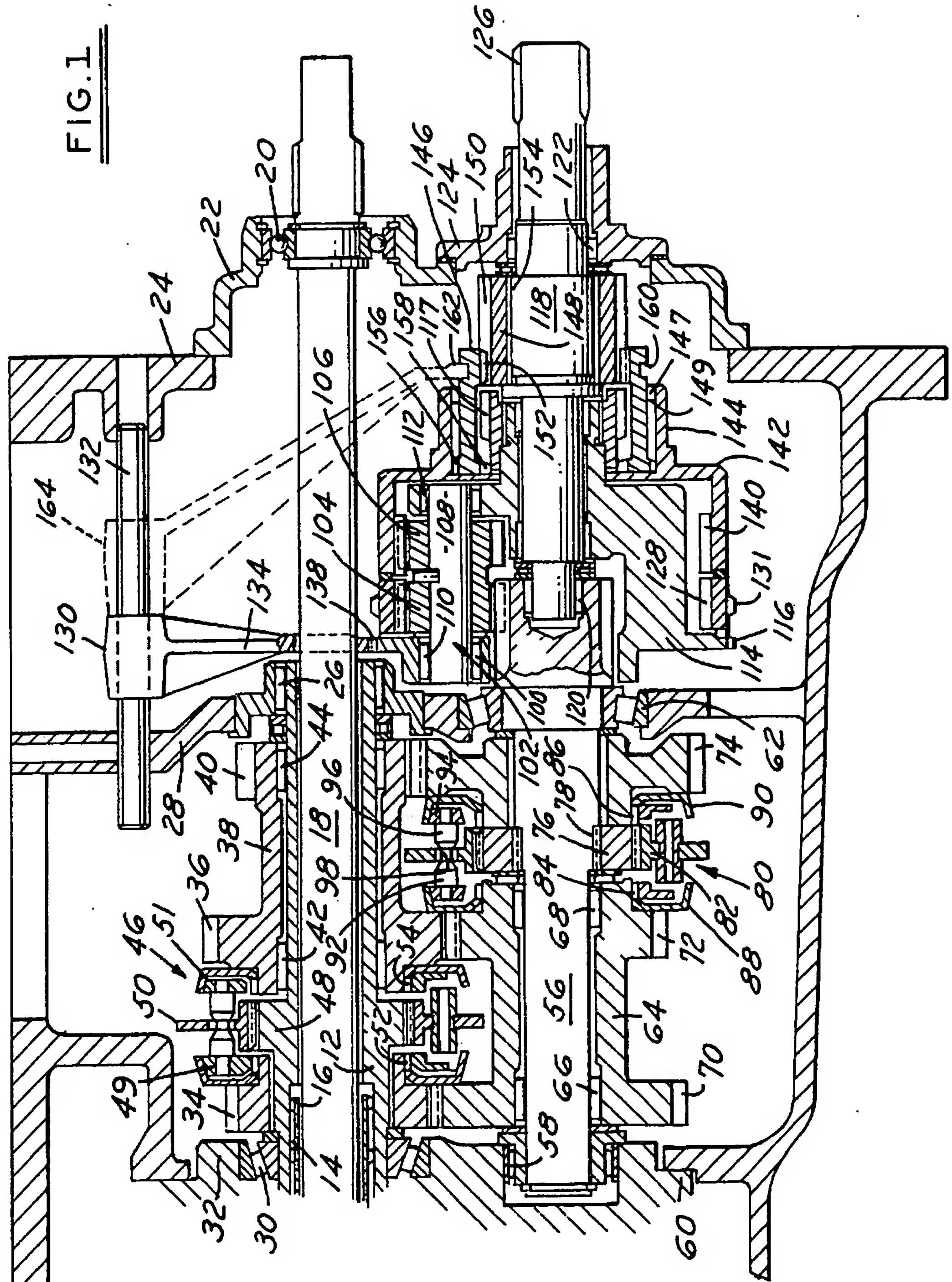
FIG.1



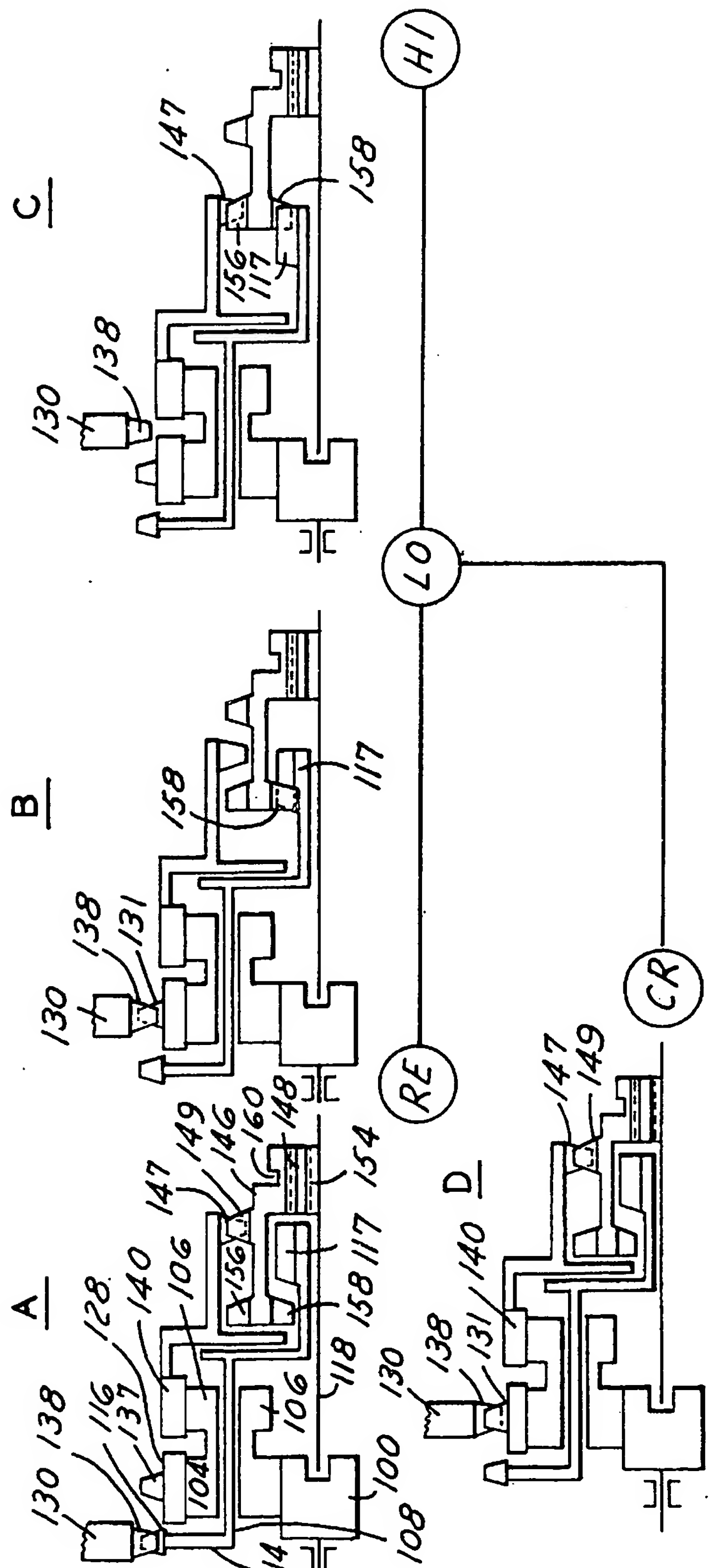

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FIG. 2

SPECIFICATION

Multiple speed range transmission

This invention relates to a multiple speed range tractor transmission and, more particularly, to a range transmission having a compound planetary gear set and a slidable clutch sleeve that selectively connects a ring gear or the planetary carrier or both of these to an output shaft.

A speed range transmission, particularly of the type for use in a tractor, may be used in combination with a synchronized transmission usually located ahead of the speed range unit, the synchronized transmission producing multiple speed ratios. Upon setting the range, which for a tractor includes reverse, low, high and a creeper range, each of the speed ratios of the synchronized transmission is available within the selected range.

This invention is an improvement of the four-speed tractor transmission described in our U.K. patent application No. 7940310.

Speed range transmissions of the type described therein require use of a first cluster gear mounted on a shaft that extends parallel to the countershaft of the synchronized transmission, the cluster assembly gear having one gear corresponding to each speed range. An output shaft aligned with the countershaft has forward drive gears and a reverse drive gear journalled on its outer surface. Between the drive gears, a selectively moveable clutch sleeve is mounted fixedly to the output shaft and moves axially into engagement with clutch teeth formed on the drive gears. Therefore, the output gear of the synchronized transmission undergoes additional torque multiplication and speed reduction according to which one of the various torque delivery paths is available upon movement of the clutch sleeves. This operation of the speed range transmission is much the same as that for producing speed ratios of the synchronized transmission; however, in shifting among the speed ranges synchronized shifting is not required.

It is desirable to reduce the manufacturing costs of a speed range unit of this type and to more compactly arrange the range unit within the transmission housing. For this reason, planetary gear arrangements driven from the synchronized transmission might be used to produce the requisite speed ranges. However, such a planetary gear system should be unsynchronized and requires selective engagement of the various driving elements without the expense and complexity of a fully synchronized power transmission.

Examples of planetary transmission adapted to produce variable speed ratios are shown in U.S. Patents 1,318,360 and 2,077,387. The shifting mechanism for related unsynchronized planetary transmission are described in U.S. Patents 4,063,470 and 4,074,591.

The range transmission according to the present invention includes a sun gear mounted for rotation about a central axis, two ring gears arranged around the sun gear, the first ring gear adapted to be selectively fixed against rotation and the second ring gear adapted to be driveably connected to the output shaft. A planetary gear set including at least one pinion cluster gear has a first planetary pinion driveably engaged with the sun gear and the first ring gear and a second planet pinion fixed to the first pinion and driveably engaged with the second ring gear. A planet pinion carrier, on which the planet pinions are rotatably mounted, is adapted to be selectively fixed against revolution about the sun gear and to be driveably connected to the output shaft. A clutch sleeve mounted for axial sliding movement relative to the pinion carrier and the second ring gear is driveably connected to the output shaft and is adapted to engage the second ring gear when positioned at a first axial location. When the clutch sleeve is moved axially to a second position, it will engage the pinion carrier. When moved to the third axial position, the clutch sleeve will driveably engage both the pinion carrier and the second ring gear. A clutch hub fixed to the output shaft provides external spline teeth that are continuously engaged with complementary teeth on the clutch sleeve so that the axial shift movement of the sleeve can be accommodated without disengagement. A first shift fork axially moveable on a shift rail is engageable with and disengageable from the pinion carrier and the first ring gear as the shift fork moves axially in relation thereto. A second shift fork mounted for axial movement on the shift rail causes the clutch sleeve to move among three axial positions.

Four speed ranges are produced with the range transmission of the invention as the clutch sleeve is moved among its three axial positions and the first shift fork moves between selected engagement with the pinion carrier, the first ring gear and an unengaged position. The range transmission simplifies the gear arrangement that would be required to produce the same result in a conventional powertrain. The clutch sleeve operates to selectively connect several elements of the gear train to the output shaft by simply moving axially in response to movement of the second shift fork. Similarly, unsynchronized engagement of the first shift fork with spline teeth machined on the outer surface of the ring gear and the pinion carrier simplifies the structure and reduces the cost and the number of elements necessary to provide four speed ranges.

The invention will now be described with reference to the accompanying drawings, in which:—

Figure 1 is a four-speed range tractor transmission comprising a cross-sectional assembly view of the principal elements of a synchronizer transmission and the multiple speed range transmission of the invention; and Figure 2 is a diagram showing the shift pattern of the four-speed range tractor tranmission and the principal elements of that transmission shown schematically in the positions they assume for each speed range. Figures 2A, 2B, 2C, 2D

illustrate changes in the driving engagements among the principal range speed transmission elements when the shift forks are moved axially among reverse drive, low range, high range and creeper range positions, respectively.

DESCRIPTION OF PREFERRED EMBODIMENTS

As shown in Figure 1, the power input sleeve shaft 12 serves as a torque input element for a multiple speed ratio synchronized transmission. Sleeve shaft 12 is splined by internal spline teeth 14 to a sleeve shaft 16 that is driven from a multi-speed transmission box (not shown) located between the synchronized transmission and the engine. A power takeoff shaft 18 extends through sleeve 12 and has one end journalled in bearing 20 located in a bearing opening formed on cone 22, which is secured to the range transmission housing 24. The end of power takeoff shaft 18 extends outwardly beyond the housing 24 to permit a driving connection with accessory tractor power implements.

Sleeve shaft 12 is journalled at its right-hand end by needle bearing 26, which is supported in an opening formed in an intermediate web 28 of the housing 24. The left-hand end of sleeve shaft 12 is journalled by tapered roller thrust bearing 30 located in bearing wall 32 of the housing 24.

A first torque input gear 34 is journalled for rotation and supported on sleeve shaft 12. A second torque input gear 36 is formed on or supported by sleeve 38. Gear 36 is part of a cluster gear assembly, which includes gear 40 formed integrally with sleeve 38 and gear 36. Sleeve 38 is supported on the sleeve shaft 12 by needle bearings 42, 44.

A double acting synchronizer clutch assembly 46 is adapted to connect selectively sleeve shaft 12 with either gear 34 or 36. The clutch assembly 46 includes a hub 48, which is externally splined to an axially movable, internally-splined clutch actuator 50. When the actuator 50 is moved to the left, the internal spline teeth formed in it drivably engage external spline teeth 52 formed on the gear 34. When shifted in a right-hand direction, to the position shown in Figure 1, actuator 50 drivably engages external splines 54 formed on the gear 36. Cone clutch elements are carried respectively by the gears 34 and 36 and rotate with them. Internal cone clutch surfaces are formed in the clutch elements which are adapted to engage external cone clutch surfaces formed on clutch elements 49 and 51. Elements 49 and 51 are connected together by cross shafts, which are formed with cam grooves that register with openings in the clutch actuator 50 when the rotation of the hub 48 is out of synchronism with one gear or the other depending upon the direction of movement of the actuator 50. The cammed edges of the groove in the cross shafts register with corresponding cam surfaces on the margins of the openings thereby creating a cone clutch engaging force that tends to force one gear or the other to rotate in synchronism with the hub 48. After synchronism is established, actuator 50 can be moved in a right-hand direction or a left-hand direction depending on the speed ratio that is desired.

A countershaft 56 is mounted in the transmission assembly in parallel disposition with respect to input shaft 12 and is journalled at its left end in bearing 58 located in a bearing opening on the support wall 60 of the housing 24. The right-hand end of the shaft 56 is supported by a roller thrust bearing 62 located in a bearing opening formed in a second portion of the interior web 28 of the housing 24.

A cluster gear assembly 64 is journalled on the countershaft 56 by bearings 66, 68. The gear assembly 64 includes a first gear element 70 and a second gear element 72, which continuously engage the gears 34 and 36, respectively. Also supported by the countershaft 56 is a drive gear 74, which meshes with gear 40.

The countershaft 56 and the hub 76, which forms a part of a second synchronizer clutch assembly 80 are splined at 78. The second synchronizer clutch assembly 80 includes an actuator 82, which can be shifted to the left or the right-hand direction depending upon the speed ratio that is desired. When actuator 82 is shifted in the left-hand direction, clutching engagement occurs between the internal splines of the actuator 82 and external splines 84 carried by the cluster gear assembly 64. When actuator 82 is shifted in the right-hand direction, it driveably engages external splines 86 carried by the gear 74. The cone clutch elements 88 and 90 are carried respectively by the gears 72, 74 and rotate with them. Internal cone clutch surfaces formed in clutch elements 88 and 90 are adapted to engage external cone clutch surfaces formed on clutch elements 92 and 94. Elements 92, 94 are connected together by cross-shafts 96. Shafts 96 are formed with cam grooves, which register with openings 98 formed in the clutch actuator 82. When rotation of the hub 76 is out of synchronism with one gear or the other depending upon the direction of movement of the actuator 82, the cammed edges of the grooves in the shafts 96 register with corresponding cam surfaces on the margins of the openings 98, thereby creating a cone clutch engaging force that tends to force the selected gear to rotate in synchronism of the hub 76. After synchronism is established, actuator 82 can be moved in a right-hand direction or left-hand direction depending upon the speed ratio that is desired.

Formed integrally with the countershaft 56 at the extreme right-hand end thereof is a sun gear 100. Three planet pinion clusters 102 are arranged symmetrically around the central axis of the countershaft 56 and extend axially parallel to the countershaft. Each pinion cluster includes a front planet pinion 104, which continually meshes with sun gear 100, and a rear planet pinion 106. Pinions 104, 106 are of unitary construction and are journalled on the outer surface of a carrier shaft 108, which is suported on bearings 110, 112 fitted within openings provided at opposite

axial ends of a planet pinion carrier 114. Carrier 114 has external spline or clutch teeth 116 extending from its outer cylindrical surface.

A torque output shaft 118 is aligned with the axis of countershaft 56 and is supported thereon at bearings 120 that are fitted within a cylindrical recess formed at the rearward extremity of the countershaft 56. The right-hand end of output shaft 118 is supported by roller bearings 122 that are received within the bearing support column 124 of the range transmission housing 24. An external spline 126 formed at the end of the output shaft 118 permits a driving connection between the shaft and a differential assembly that transmits output torque to the driving wheels of the vehicle. Planet pinion carrier 114 is loosely fitted over the outer surface of the output shaft 118.

A front ring gear 128 is continuously in meshing engagement with the front planet pinion 104. External spline teeth 130 are formed on the outer cylindrical surface of ring gear 128 in axial alignment with the spline teeth 116 of the planet pinion carrier 114.

A first shift fork 130 is slidably mounted on a shift rail 132 and has an arm portion 134 that extends radially downward straddling the power takeoff shaft 18 and terminating in an arcuate surface 138 that has internal spline teeth formed therein complementary to the external spline teeth 116, 130 of the planet carrier 114 and front ring gear 128.

A rear ring gear 140 is located behind the front ring gear 128 and is held in meshing engagement continuously with the rear planet pinion 106. Formed integrally with ring gear 140 is a disk portion 142 and a sleeve portion 144 that extends axially generally parallel to the axis of the output shaft. Sleeve portion 144 has internal spline teeth 147 formed around its inner surface.

A clutch sleeve 146 is slidably supported on and drably connected to a clutch hub 148, which is loosely fitted over the outer surface of the output shaft 118. External clutch teeth 150 formed on the outer surface of the clutch hub 148 engage internal clutch teeth 152 formed on the inner surface of the clutch sleeve 146. Clutch hub 148 is drivably fixed at spline 154 to the output shaft 118. External spline teeth 156 and internal spline teeth 158 are formed at the forward end of clutch sleeve 146. External teeth 156 are complementary to and can engage the internal spline teeth 147 of the rear ring gear 140 as the clutch sleeve is moved axially toward the rear of the transmission housing. Similarly, internal spline teeth 158 of the clutch sleeve 146 can engage spline teeth 117 of the planet pinion carrier 114 as the clutch sleeve 146 is moved axially to the rear. Clutch sleeve 146 has an annular recess 160 that is continuously engaged by the end 162 of a second shift fork 164 slidably mounted on the outer surface of the shift rail 132 behind the first shift fork 130.

To establish low speed ratio operation of the synchronized multi-speed transmission, actuator 50 is shifted to the left direction and the actuator 82 is shifted in the right-hand direction. Both synchronizers must be engaged to effect a complete torque flow path through the gearing. When the synchronizers are engaged in this fashion, torque is delivered from the multi-speed transmission by sleeve shaft 16 through the synchronizer clutch hub 48 to gear 34. This action drives the cluster gear assembly 64, which causes the second gear element 72 to drive the second input gear 36 and gear 40 to drive gear 74. Power is transmitted through the engaged clutch spline 86 and the actuator hub 76 to the countershaft 56.

A ratio change from the first speed ratio to the second speed ratio requires actuation only of synchronizer clutch 46; synchronizer clutch 80 remains in the position that is assumed during first speed ratio operation. Actuator 50 is shifted in a right-hand direction thereby establishing a driving connection between the hub 48 and the gear 36. The same torque flow path exists although the driving gear is gear 36 instead of gear 34. Thus, the overall ratio is increased because of the larger pitch diameter of the gear 36.

A speed ratio change from the second speed ratio to the third speed ratio requires actuation of both synchronizer clutches. To makes this speed ratio change, synchronizer clutch assembly 46 is first moved to the neutral position shown in Figure 1, then synchronizer clutch assembly 80 is shifted leftwardly thereby establishing a connection between the countershaft 56 and the gear 72. After the shift is completed, synchronizer clutch assembly 46 is shifted to the left thereby connecting gear 34 to the torque input sleeve shaft 16. During this ratio change the synchronizer clutch assembly 80 establishes synchronism among the torque delivery elements while the inertia of the power unit and the input clutch (not shown) is disconnected from the rotating mass thus facilitating synchronization. The same reduced inertia shift occurs when the transmission mechanism shifts to a lower ratio, specifically from the third ratio to the second ratio. In that case, the synchronizer clutch 46 is moved initially to the neutral position, thereby disconnecting the rotating inertia on the torque input side of sleeve shaft 16 as the synchronizer clutch assembly 80 again establishes a synchronized driving connection between the countershaft 56 and the cluster gear assembly 64.

A ratio change from the third speed to the high speed, or fourth drive ratio, is achieved by maintaining synchronizer clutch 80 in the left-hand position, which it assumed during third speed operation. It is necessary to actuate only synchronizer clutch assembly 46. Actuator 50 is moved in a right-hand direction thereby establishing a driving connection between sleeve shaft 16 and gear 36. Torque is then delivered from the input shaft through clutch assembly 46 to gears 36 and 72 through clutch assembly 82 and to countershaft 56.

Operation of the four-speed range transmission

is demonstrated schematically in Figure 2. The range transmission can produce three forward speed ranges, high speed, low speed, and creeper speed, and one reverse speed ratio.

Reverse drive results when both shift forks 130, 164 are actuated as a single unit. When the forks are positioned at the left end of shift rail 132 the transmission is disposed for reverse speed operation. The internal spline teeth 138 of the first shift fork 130 engage external spline teeth 116 of the planet pinion carrier 114 thereby locking the carrier against revolution about the axis of the input shaft. Clutch sleeve 146 is moved to its extreme forward position as shown in Figure 2A causing its spline teeth 149 to driveably engage the spline teeth 147 of the ring gear 140. For reverse operation, power is transmitted from the input sun gear 100, through the planet cluster pinions 104, 106, which rotate on the carrier shaft 108 but do not revolve, to the ring gear 140, which drives clutch sleeve 146, hub 148 and finally to output shaft 118.

Low range operation results when shift forks 130, 164 are moved to the middle position of the shift rail 132. This causes spline teeth 138 to move out of engagement with the teeth 116 of planet carrier 114 and into driving engagement with spline teeth 131 formed on the front ring gear 128. This engagement causes the ring gear to be locked against rotation and furnishes a fixed element on which planet cluster 114 revolves. Forward movement of the second shift fork 164 moves hub 148 forward causing spline teeth 149 to disengage spline 147 and the spline 158 of clutch sleeve 146 to drivably engage the external spline teeth 117 of the planet pinion carrier 114. With the shift elements so positioned, the torque delivery path includes input sun gear 100, front planet pinion 104, planet carrier 114, clutch sleeve 146, clutch hub 148 and output shaft 118.

The high speed range is produced when shift forks 130, 164 are moved on shift rail 132 to the extreme rearward position. This causes spline teeth 138 of shift fork 130 to become disengaged. Rearward movement of clutch sleeve 146 causes its external spline teeth 156 to engage the internal splines 147 of the ring gear 140 but allows its internal spline teeth 158 to remain drivably engaged with spline teeth 117 formed on the planet pinion carrier 117. This action locks the planetary gear set 102 directly to the output shaft 118, which is driven directly from the input sun gear 100, and the range speed portion of the transmission produces a high speed ratio.

An additional low speed range, creeper speed, which is approximately one-eighth the speed of the low range, is produced when the shift forks 130, 164 are operated independently. To produce creeper speed range, shift fork 130 is placed in the intermediate, or low speed ratio range position, and shift fork 164 is shifted to the extreme rearward position on rail 132. This action establishes a driving connection between rear ring gear 140 and the output shaft 118, as was done with the high speed ratio range, by causing splines 147 and 156 of ring gear 140 and clutch sleeve 146, respectively, to become engaged. Again, shift fork 130 locks the planetary gear set against revolution about the central axis of the input shaft by causing spline 138 to engage the external splines 131 of ring gear 128. The torque flow path with the range transmission disposed for a creeper speed range, therefore, includes input sun gear 100, the compound planetary gear set 102, clutch sleeve 146, clutch hub 148 and output shaft 118.

CLAIMS

1. A multiple speed range transmission comprising:

a sun gear mounted for rotation about a central axis;

an output shaft;

first and second ring gears disposed about the sun gear axis, said first ring gear adapted to be selectively fixed against rotation, said second ring gear adapted to be drivably connected selectively to said output shaft;

a planetary gear set including at least one pinion cluster gear having a first planet pinion drivably engaged with said sun gear and said first ring gear, a second planet pinion fixed to said first pinion and drivably engaged with said second ring gear;

a planet pinion carrier on which the planet pinions are mounted for rotation about their respective axes, adapted to be selectively fixed against revolution about the axis of said sun gear and to be drivably connected selectively to said output shaft; and

means for selectively fixing said pinion carrier and said first ring gear against rotation.

2. The speed range transmission according to Claim 1 further comprising means for selectively connecting said planet pinion carrier and said second ring gear to said output shaft.

3. The speed range transmission according to Claim 2 wherein said connecting means includes a clutch sleeve mounted for axial sliding movement relative to said pinion carrier and said second ring gear, drivably connected to said output shaft, adapted to engage drivably said second ring gear when said clutch sleeve is located at a first axial position, adapted to engage drivably said pinion carrier at a second axial position and adapted to engage drivably said pinion carrier and said second ring gear at a third axial position.

4. The speed range transmission according to Claim 3 wherein said second ring gear and said pinion carrier have spline teeth formed thereon, said clutch sleeve further including a first set of spline teeth engageable with and disengageable from the spline teeth engageable with and disengageable from the spline teeth of said second ring gear as said clutch sleeve is more axially, a second set of spline teeth engageable with and disengageable from the spline teeth of said pinion carrier as said clutch sleeve is moved axially and a third set of spline teeth engageable with and disengageable from the spline teeth of said second ring gear as said clutch sleeve is

moved axially.

5. The speed range transmission according to Claim 3 or Claim 4 wherein said clutch sleeve has a recess formed therein and further including a second shift fork mounted on a shift rail for axial sliding movement having one end engaging the recess formed on said clutch sleeve whereby said clutch sleeve is moved axially as said second shift fork moves.

6. The speed range transmission according to any one of Claims 3 to 5 further including a clutch hub fixed to said output shaft having spline teeth continuously engaged by a fourth set of spline teeth formed on said clutch sleeve as said sleeve is moved axially, whereby a driving connection between said output shaft and said clutch sleeve is made.

7. The speed range transmission according to any one of the preceding claims wherein said fixing means comprises: a shift rail; a first shift fork mounted on said shift rail for axial sliding movement having spline teeth engageable with and disengageable from a second set of spline teeth formed on said pinion carrier as said first shift fork is moved axially relative thereto, spline teeth formed on said first ring gear, the spline teeth of said first shift fork being engageable with and disengageable from the spline teeth of said first ring gear as said clutch sleeve is moved axially relative thereto.

8. A multiple speed range transmission substantially as hereinbefore described with reference to and as shown in the accompanying drawings.

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